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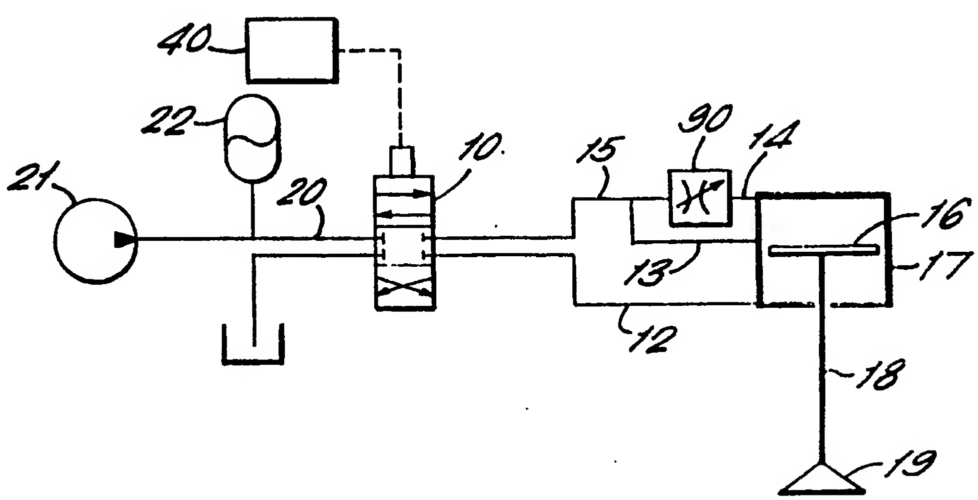

(57) Abstract

The present invention relates to valve control apparatus for use in an engine or compressor to open and close an inlet or exhaust port thereof which comprises valve means (19), an irreversible hydraulic actuator (11) and control means (10, 40). The actuator (11) is connected to the valve means (19) such that the valve means (19) can only operate when driven by the actuator (11). The actuator (11) is irreversible since the piston (16) with the actuator (11) cannot move unless fluid is introduced into one of the chambers of the actuator and removed from the other.

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VALVE CONTROL APPARATUS

The invention relates to valve control apparatus for use in engines or compressors.

The present invention will be discussed with reference to its use in a reciprocating internal combustion engine. However, the invention should not be considered limited to such. The invention could be used to actuate valves in any type of engine or any form of compressor.

The motion of valve gear at the cylinder head of a reciprocating I.C. engine is normally controlled by the rotation of one or more cam shafts. The cam shafts act to open and close the inlet and exhaust valves of the internal combustion engine. The timing of the opening and closing of the exhaust valves is determined by the profile of the cam shaft. Different cam shaft profiles are chosen for different engines. The profile of the cam shaft has a large impact on the efficiency and performance of an engine.

In an ideal engine the inlet and exhaust valves would either be opened or closed, with no intermediate position. Having a valve in an intermediate position between opening and closing is not ideal, since unnecessary resistance is placed upon the flow of gases into and out of the working cylinders of the engine. However, when cam shafts are used, the valve can only be accelerated towards and away from the valve seat up to a certain level of acceleration. Beyond a certain level the valve gear at the cylinder head loses its integrity and the motion of the valve

cannot be controlled. In a system where the cam shaft compresses a rocker arm which presses on the top of the valve stem, if the cam shaft decelerates the rocker arm too quickly, the rocker arm may become detached from the valve stem, due to the inertia of the valve stem.

Systems have been proposed in the past which provide an alternative to the activation of valve gear by cam shafts. Such an alternative can be found in U.K. Patent GB 20061560B in the name of Daimler-Benz. In the system described in GB 20061560 a valve element is connected to a working piston which can move within a cylinder. The position of the valve element is varied by the movement of the working piston. The working piston is subjected to a pre-tensioning by a spring which biases the piston to move in one direction. The movement of the piston is controlled by the supply of pressurised medium into a chamber defined by one surface of the piston and the inner surface of the cylinder. The force acting on the piston due to presence of pressurised medium in the chamber acts against the resilient means. An electronic controller is used to control the supply of the pressurised medium to the chamber and thereby controls the movement of the valve element.

In U.S. Patent 4188925 the valve gear at the cylinder head is controlled by the supply of pressurised medium to an actuator which pushes against the top of the valve stem, against the action of a spring. A control system controls the supply of pressurised fluid to control the motion of the valve.

In U.S. 4612883 once again a system is described in which a valve is controlled by supplying

pressurised fluid to one side of the piston, the force applied by the pressurised fluid acting against a biasing force supplied by a spring.

The present invention is an improvement in the prior art since the hydraulic system used to control the valve motion in the present invention does not contain any passive elements such as springs. The motion of the valve is instead controlled by the control of the supply of pressurised medium to both sides of a piston to which the valve is attached. The system enables the motion of the valve to be controlled in all positions and allows for controlled acceleration and decceleration of the valve in both possible directions of motion of the valve.

The present invention provides valve control apparatus for controlling valve means in an engine or a compressor to open and close an inlet or exhaust port thereof comprising an irreversible hydraulic actuator connected to the valve means such that the valve means can only operate when driven by the actuator and control means for controlling the actuator.

References to an irreversible hydraulic actuator in this specification are references to an actuator which comprises a piston and cylinder arrangement and in which the position and motion of the piston are controlled in both directions of motion by control signals received by the actuator, flow of fluid into one chamber with corresponding flow of fluid out of the second chamber of the actuator being necessary to effect motion.

Preferably the control means comprises signal

generating means and control valve means controlled by a signal generated by the signal generating means, which control valve means control the flow of hydraulic fluid to and from the irreversible hydraulic actuator.

Preferably the signal generating means generates signals having regard to the stroke position of the engine or compressor and in accordance with preprogrammed instructions.

Preferably the signal generating means modifies the generated signal having regard to the operational speed of the engine or compressor.

Preferably the irreversible hydraulic actuator comprises resistance means, which resistance means damp the motion of the piston of the actuator as it approaches at least one of its limits of motion.

The present invention also provides valve control apparatus for controlling valve means in an engine or compressor to open and close an inlet or exhaust port thereof comprising:

an actuator which comprises a piston within a cylinder, which piston is connected to the valve means such that motion of the piston causes motion of the valve means;

measuring means for measuring the stroke position of the engine or compressor; and

control means for controlling the flow of fluid or gas into both of the chambers defined by the surfaces of piston and the interior surface of the cylinder;

wherein the control means controls the flow of fluid or gas to cause the valve means to open and

close the inlet or exhaust port and wherein the control means controls the functioning of the valve means having regard to the stroke position of the engine or compressor and in accordance with a set of preprogrammed instructions.

Preferably the valve control apparatus further comprises position measurement means for measuring the position of the actuator piston with respect to the actuator cylinder and wherein the control means controls the functioning of the valve means with regard to the piston position.

Preferably the control means of the apparatus comprises;

processing means which produces a control signal;

a source of pressurised fluid or pressurised gas;

an exhaust for pressurised fluid or pressurised gas; and

control valve means which can connect each of the chambers defined by the piston surfaces and the interior surface of the cylinder to the source of pressurised fluid or gas or to the exhaust for pressurised fluid or gas; and

wherein the controlled valve means are controlled by the control signal produced by the processing means.

Preferably the measuring means for measuring the stroke position of the engine or compressor produces an electrical signal indicative of the stroke position, which electrical signal is relayed to the processing means.

In a preferred embodiment the processing means includes calculating means which calculates from the

stroke position signal the rotational speed of the engine and the processing means controls the operation of the valve means with regard to the engine rotational speed.

The controlled valve means preferably comprises a servo-valve.

The invention further provides an internal combustion engine having at least one working cylinder with an inlet or an exhaust port controlled by the engine valve control apparatus of the invention and also a vehicle having such an engine.

The invention can be used in an internal combustion engine to throttle the engine by connecting actuators to the inlet valves of the working cylinders of the engine providing a sensor to generate a signal indicative of engine speed demand by the user, which signal is processed by the control means.

Preferred embodiments of the present invention will be described with reference to the accompanying drawings in which;

Figure 1 is a part cross-section through a portion of the apparatus including the hydraulic actuator and the control valve.

Figure 2 is a schematic representation of a portion of the apparatus including a pump, a reservoir, a control valve, the hydraulic actuator and the engine valve.

Figure 3 is a schematic representation of the control system of the apparatus.

Figure 4 is a cross-section of part of the hydraulic actuator of the apparatus.

Figure 5 is a schematic representation of a

second embodiment of the apparatus.

Figure 6 is a cross-section of part of the hydraulic actuator of the second embodiment of the apparatus.

Referring to Figure 1 the apparatus can be seen to comprise a control valve 10 and an irreversible hydraulic actuator 11 connected to the control valve 10 by three passages 12, 13 and 14. Passages 13 and 14 are joined together at a point 15.

The irreversible hydraulic actuator 11 comprises a piston 16 movable within a cylinder defined by the walls 17. The piston 16 is directly connected by a rod 18 to a valve (not shown in Fig. 1) which opens and closes an aperture opening onto a cylinder of an internal combustion engine.

The apparatus also includes a displacement measurement device 19, such as an LVDT (Linear Variable Differential Transformer) or a Hall Effect Transducer, connected to the piston 16 for measurement of the relevant displacement of the piston 16 with respect to the cylinder 17.

The control valve 10 is a standard control valve. In the preferred embodiment in a engine with a maximum revolutionary speed of 7,000 rpm the valve has a frequency response of 350 Hz.

The arrangement can be seen schematically in Figure 2. The hydraulic actuator 17 is shown schematically as a cylinder comprising a piston 16 connected by the rod 18 to the valve 19. The piston 16 is caused to move with respect to the cylinder 17 by supplying fluid under pressure through the two

pipes 15 and 12 to both sides of the piston 16.

The control valve 10 is shown schematically in Figure 2 with two control ports connected to lines 15 and 12, a pressure port connected to line 20 and a return port connected to line 25. The pressure port 20 is connected via the line 20 to a pump 21 and a reservoir 22. The control valve 10 supplies pressurised fluid to either line 15 or 12 depending upon an input signal received from via the line 26 from the processing means 40. When one line is so connected the other line is connected by the control valve 10 to the exhaust 24 via the line 25.

An expanded cross-sectional view of part of the hydraulic actuator 11 is shown in Figure 4. In Figure 4 there can be seen the two tubes 13 and 14 the rod 23 of the displacement measurement device 19, the piston 16 and the cylinder defined by the walls 17. Circular seals 27, 28 and 29 are provided to prevent hydraulic fluid from passing from one compartment to another.

The Figure 4 shows clearly an arrangement which forms the resistance means of the invention. The purpose and method of operation of the resistance means shall be hereinafter described. The resistance means are shown schematically at 90 in Figure 2. The resistance means of the first embodiment of the invention comprises an annular ring of deformable resilient material 31, an annular washer 32 and a sleeve 33. The sleeve 33 surrounds the rod 23 of the displacement measurement device 19. The sleeve 33 has a series of apertures 34 therein which allow communication of the inside of the sleeve with the outside thereof. Another important integer in the resistance means is an orifice formed by a nozzle 35.

The control system of the apparatus can be seen in Figure 3. The processing means of the apparatus is contained within the dotted line 40. From Figure 3 it can be seen that there are three inputs to the processing means. The first of these inputs is an electrical signal corresponding to the crank angle. The crank angle is measured at 41. The crank angle signal corresponds to an instantaneous measurement of the angle that the crankshaft makes with an arbitrary fixed reference. The crank angle measurement is used to provide an indication of the stroke position of the engine, in other words the relative displacement of the working pistons with respect to their working cylinders. Many other measurements could be made to provide a signal indicative of stroke position. For instance measurements could be taken from the timing mechanism of the engine.

The instantaneous stroke position measurement is differentiated with respect to time at 42 to give the speed of revolution of the engine in revolutions per minute. This parameter forms another input into the processing means 40. The crank angle measurement and the RPM measurement are used not only by the processing means 40 but can also used by processing means for controlling ignition timing and fuel injection.

The third input to the processing means 40 is a position measurement. The displacement measurement means 19 measures the position of the piston 16 with respect to the cylinder formed by the walls 17. The position measurement is input to the processing means. The signal is used in the closed loop control system of the processing means 40, the signal

providing the necessary feedback loop.

The processing means 40 has one output 43. The output is an electrical signal which controls motion of the control valve 10. The control valve 10 then controls the motion of the piston 16.

The method of operation of the processing means 40 will be hereinafter described, once the physical operation of the actuator 11 has been described.

The method of operation of the control apparatus will now be described with reference to the drawings and with reference to the preceding description of the drawings.

Referring firstly to Figure 2 of the drawings, the piston 16 can move within the cylinder 17 and is connected via a rod 18 to a valve 19. The valve 19 opens and closes an aperture which opens on to a cylinder of the internal combustion engine. The valve can shut either an inlet aperture opening on to the cylinder or an exhaust aperture.

The piston 16 is caused to move within the cylinder 17 by applying a pressure difference thereacross. The pressure difference is applied by supplying hydraulic fluid under pressure to one side of the piston 16, whilst connecting the hydraulic fluid on the other side of the piston 16 to a sink 24.

The control valve is used to control the motion of the piston 16. If the servo-valve 10 is used to cause the valve 19 to move towards the valve seat then the servo-valve acts to connect the line 12 to line 20. Line 20 receives a supply of hydraulic fluid

under pressure from a pump 21 and a reservoir of pressurised hydraulic fluid 22.

Whilst the control valve 10 connects the line 12 to the line 20 to supply hydraulic fluid under pressure to the lower side of the piston 16, the valve also acts to connect line 15 to a sink 24. Sink 24 is a sink for hydraulic fluid and contains fluid at low pressure. By connecting the line 15 to the sink 24 the control valve enables fluid to flow from the upper portion of the cylinder 17 through the line 15 and out to the sink 24.

The reservoir of pressurised hydraulic fluid 22 is used to maintain the supply pressure at 20 at an approximately constant value. The pump 21 can be powered by motion of the engine which varies during use of the vehicle and therefore the supply of hydraulic fluid from pump 21 can vary. The reservoir 22 contains a supply of pressurised fluid and acts to stabilise the fluctuations in supply of pressurised hydraulic fluid.

The reservoir 22 also helps to offset problems caused by inertia of the hydraulic fluid. When the control valve 10 opens to require the supply of pressurised hydraulic fluid, a large counter fluid is required quickly. The hydraulic fluid has inherent inertia. If the supply is restricted to that available in a supply line, pump then the inertia of the fluid may cause inaccuracies in control, since insufficient fluid may be readily available for supply to the actuator. The reservoir 22 offsets this difficulty by providing a ready supply of pressurised fluid. In practice, the reservoir 22 should be situated as closely as possible to the control valve

10.

In practice the apparatus of the invention has been found to be limited in performance by the frequency response of the control valve 10 and the position measurement means 19. The control system is not sensitive enough to control the decceleration of the engine valve onto the seat of the cylinder head. The problem is enhanced since the fluid contained within the hydraulic actuator is not a perfectly incompressible fluid. The fluid tends to compress under rapid decceleration of the valve. This compression causes inaccuracies in control. Further inaccuracies are caused when the fluid tries to expand, applying pressure to the piston. In effect the compressibility of the fluid within the hydraulic actuator causes "bouncing" of the controlled valve. In order to overcome these difficulties the apparatus includes a resistance means the operation of which shall now be described.

The resistance means only operates during the last portion of the upward stroke of the piston, as the valve approaches the valve seat. As the piston 16 moves away from the engine to bring the valve 19 into contact with its respective valve seat, the non-deformable washer 32 contacts the lower end of the sleeve 33. Before such contact hydraulic fluid may flow out of the cylinder 17 through the outlet 14 by passing within the sleeve 33 and through the holes 34 which communicate with an annular space defined within the actuator. When the washer contacts with the sleeve 33 the resilient components 31 deforms to seal against the rod 23 of the displacement measurement means 19. The seal prevents further hydraulic fluid from passing out of the cylinder 17 through the sleeve

33 to the outlet tube 14. Further fluid can only be expelled from the upper portion of the cylinder 17 through the tube 13.

A nozzle 35 is positioned within the tube 13 to provide a restricted aperture. The restricted aperture provides resistance to flow of fluid out of the cylinder 17 as the piston 16 approaches the end of its upward stroke. This resistance effectively damps the motion of the piston 16, slowing the associated motion of the valve towards the valve seat, slowing the valve so that impact between the valve and the valve seat does not occur.

The resistance means also largely eliminates the problem of the valve "bouncing" on the hydraulic fluid within the actuator since the volume of fluid enclosed between the piston 31 and the orifice defined by the nozzle 35 is greatly reduced in comparison to the volume of fluid usually subject to compressing forces. The volume of enclosed fluid must be kept as small as possible to avoid any difficulties caused by the resonant frequency of the enclosed fluid.

After the piston has reached the top of its upward motion and fluid under pressure is supplied to the top half of the piston through the inlet 14 and 13 the resistance means ceases to function, since the hydraulic fluid supplied through 14 can apply pressure to the washer 32 and the piston 16. Unnecessary damping of the motion of the piston 16 within the cylinder 17 is therefore avoided.

In alternative arrangement of the resistance means will now be described with reference to Figures 5 and 6. In Figure 5 there can be seen an alternative

embodiment of an actuator 100 for use in the apparatus. The actuator 100 comprises a piston 101 attached by a connecting rod 102 to an inlet or exhaust valve of an engine 19.

The actuator 100 has one fixed end 103 and one movable end 104. The movable end 104 is connected to a piston 105. In the preferred embodiment described the movable end 104 is formed in one component with the piston 105. The piston 105 is movable within a second cylinder 106. A cavity 107 is defined between the surface 108 of the piston 105 and the uppermost surface 109 of the second cylinder 106.

Hydraulic fluid is supplied to the chamber 107 by a line 110. Positioned in the line 110 is an orifice 111. The resistance means 112 of the second preferred embodiment of the invention is shown schematically in Figure 1 in the line 113. Line 113 connects the chamber 114 of the actuator 100 to the valve 115. A line 116 connects the lower chamber 117 of the actuator 100 to the valve 115.

The valve is further connected by a line 118 to a source of hydraulic fluid under pressure 119 and a reservoir of pressurised hydraulic fluid 120. The control valve 115 is also connected by a line 121 to an exhaust for fluid 122. The control valve 115 is controlled by electrical signals received through a line 123 from the processing means 40 of the invention.

The chamber 107 of the second cylinder 106 is connected to both the source of pressurised fluid 119 and the reservoir of pressurised fluid 120 via the line 110. The line 110 is not directly connected to the source of pressurised fluid 119 and the reservoir

of pressurised fluid 120. Instead, the line 110 is connected to both the source 119 and the reservoir 120 via a regulating valve 124. The regulating valve 124 regulates the pressure of the fluid in line 110 to one tenth of the system pressure.

In Figure 6 there can be seen a part cross-section of the actuator 100. In Figure 6 there can be seen the movable end 104 connected to the piston 105, as well as the second cylinder 106, the chamber 107 and the supply line 110.

The resistance means of the second embodiment of the invention comprises the essential integers of a frusto-conical protrusion 125 on the lower surface of the movable end 104 and also the frusto-conical indentation 126 on the upper surface of the piston 101.

In normal operation of the actuator 100, as the piston 101 approaches the end 104 fluid flows freely out of the chamber 114 to the line 113 via cavities 127 and 128 in the movable end 104. However, as the piston 101 approaches the movable end 104 the flow of fluid out of the actuator 100 to the line 113 becomes constricted. The lower surface of the movable end 104 and the upper surface of the piston 101 define an annular cavity 115. As the piston 101 moves towards the movable end 104, the flow of fluid out of the region 115 becomes restricted due to the narrow nature of the annular aperture 116 defined between the protrusion 125 and the uppermost surface of the piston 101.

As the piston 101 approaches nearer and nearer the movable end 104,the greater becomes the resistance offered to the flow of fluid out of the region 115.

This arises since the constriction formed between the uppermost piston surface and the lowermost movable end surface becomes narrower and narrower. The progressively increasing resistance offered to the flow of fluid out of the actuator 100 progressively damps the motion of the piston 101 towards the movable end 104. The progressively increasing resistance allows progressive decelaration of the motion of the valve 19 towards its associated valve seat. Unlike the first embodiment described above, the resistance means of the second embodiment progressively increases the resistance to the flow of fluid out of the actuator 100 and hence progressively increases the deceleration forces on the valve 19. In the first embodiment described above, the resistance offered to the flow of fluid out of the actuator 11 is determined by the orifice formed in the nozzle 35.

The second embodiment of the invention relies upon the fact that very little or no fluid is left between the piston 101 and the movable end 104 when the valve 19 is against its respective valve seat. Therefore adjustment means is provided which allows the movable end 104 to be brought into contact with the piston 101 when the valve 19 is against its respective valve seat.

The adjustment means comprises the line 110, the orifice 111 and the pressure regulating valve 124. When the pressure regulating valve 124 is operated, a supply of fluid under pressure is provided to the chamber 107 of the second cylinder 106. The flow of fluid into the chamber 107 is kept to a very low level by use of the orifice 111. By the introduction of fluid into the chamber 107 the piston 105 causes the end 104 to move down to mate with the piston 101.

Motion of the movable end 104 is kept to a minimum during operation of the actuator 100, since the nozzle 111 restricts any flow of fluid into or out of the chamber 107. Further, the compressibility of the fluid in the chamber 107 has negligible effects, since the piston 105 acts over a comparatively large area, whilst the volume of fluid contained within the chamber 107 is small.

In the second embodiment hereinbefore described position measurement means for measuring the displacement of the actuator piston is not shown. However, such means can be easily provided. Alternatively the position of the piston can be determined by the processing means 40 by suitable integration of the control signals generated. As another option, the flow of gases into the associated working cylinder of the engine could be monitored. Such gas flow is proportional to inlet valve lift and the inlet valve lift would correspond to the piston position of the relevant hydraulic actuator.

The method of operation of the processing means 40 of the apparatus shall now be discussed, with reference to Figure 3.

As mentioned before, signals corresponding to the crankshaft position and to the speed of revolution of the engine are input into the processing means. The signals are then processed using a "look-up table" with linear interpolation, in box 45. The "look-up table" is programmed into apparatus by the operator. The operator can in this way instruct the system to follow a particular cam shaft profile.

A position signal for the valve is input into a summing junction at 46 and is summed with the actual valve position as measured by the displacement measuring means. The resulting error signal is fed into a PID (Proportional, Integral, Differential) control system at 47. The output of the PID 47 is fed into a decision box 48. The decision box decides whether the control system requires the valve to move towards or away from the valve seat. In mathematical terms the decision box decides whether the drive signal is greater or less than zero. Depending upon whether the drive signal is greater or less than zero the processing means multiplies the drive signal by a constant GF or GB. This constant converts the position control signal into a signal controlling the servo-valve. Different scaling factors GF and GB are needed since the piston 16 has different surface areas on either side thereof, since it is attached to the rod 18 on one side and is attached to the rod 23 on the other side.

The apparatus of the invention controls the motion of the actuator piston in both directions. Since the apparatus contains no resilient means the valve lift can be calculated by the processing means from the position of the actuator piston. Hence the system can be used to model the motion of valves operated by mechanical means which use a cam shaft with a particular cam profile. Alternatively the system could be used to replace the conventional throttles used in automobile engines. The system could limit inlet valve lift to control the amount of fuel/air mixture flowing into a working cylinder and thereby acts as a throttle.

The system is advantageous over the ordinary cam

shaft system since the valve can be accelerated and decelerated without the limits imposed by mechanical linkages.

Whilst the system described above uses pressurised hydraulic fluid, the applicant envisages that compressed gas could also be used to cause motion of the piston 16. However, the use of hydraulic fluid is suited to the use of the apparatus in an internal combustion engine since pressurised hydraulic fluid is readily available in the form of oil pressurised by an oil pump powered by the output of the engine.

Whilst the system is described above in use in a reciprocating internal combustion engine the invention could obviously be used in a reciprocating compressor, or, indeed, any type of engine which uses valves to control flow of fluids of gases.

It will be clear from the foregoing that the invention enables accurate control of valve motion. In the systems of the prior art the use of passive elements introduces inaccuracies in control of the valve. The present invention is an improvement over the prior art since it eliminates the need for reactive elements such as springs. The present invention provides a system which can fully control the motion of a valve.

CLAIMS

1. Valve control apparatus for controlling valve means in an engine or compressor to open and close an inlet or exhaust port thereof comprising

an irreversible hydraulic actuator connected to the valve means such that the valve means can only operate when driven to move by the actuator and

control means for controlling the actuator.

2. Valve control apparatus as claimed in Claim 1 wherein the control means comprises signal generating means and control valve means controlled by a signal generated by the signal generating means, which control valve means controls the flow of hydraulic fluid to and from the irreversible hydraulic actuator.

3. Valve control apparatus as claimed in Claim 2 wherein the signal generating means generates signals having regard to the stroke position of the engine or compressor and in accordance with preprogrammed instructions.

4. Valve control apparatus as claimed in Claim 3 wherein the signal generating means modifies the generated signal having regard to the operational speed of the engine or compressor.

5. Valve control apparatus as claimed in any one of the preceding claims wherein the irreversible hydraulic actuator comprises resistance means, which resistance means damp the motion of the piston of the actuator as it approaches at least one of its limits of motion.

6. Valve control apparatus for controlling valve means in an engine or compressor to open and close an inlet or exhaust port thereof comprising:

an actuator which comprises a piston within a cylinder, which piston is connected to the valve means such that motion of the piston causes motion of the valve means;

measuring means for measuring the stroke position of the engine or compressor; and

control means for controlling the flow of fluid or gas into both of the chambers defined by the surfaces of piston and the interior surface of the cylinder;

wherein the control means controls the flow of fluid or gas to cause the valve means to open and close the inlet or exhaust port and wherein the control means controls the functioning of the valve means having regard to the stroke position of the engine or compressor and in accordance with a set of preprogrammed instructions.

7. Valve control apparatus as claimed in Claim 6 comprising additionally position measurement means for measuring the position of the actuator piston with respect to the actuator cylinder and wherein the control means controls the functioning of the valve means with regard to the piston position.

8. Valve control apparatus as claimed in Claim 6 or Claim 7 wherein the control means comprises;

processing means which produces a control signal;

a source of pressurised fluid or pressurised gas;

an exhaust for pressurised fluid or pressurised gas; and

control valve means which can connect each of

the chambers defined by the piston surfaces and the interior surface of the cylinder to the source of pressurised fluid or gas or to the exhaust for pressurised fluid or gas; and

wherein the controlled valve means are controlled by the control signal produced by the processing means.

9. Valve control apparatus as claimed in Claim 8 wherein the measuring means for measuring the stroke position of the engine or compressor produces an electrical signal indicative of the stroke position, which electrical signal is relayed to the processing means.

10. Valve control apparatus as claimed in Claim 9 wherein the processing means includes calculating means which calculates from the stroke position signal the rotational speed of the engine and the processing means controls the operation of the valve means with regard to the engine rotational speed.

11. Valve control apparatus as claimed in any one of Claims 8 to 10 wherein the controlled valve means comprises a servo-valve.

12. An internal combustion engine having at least one working cylinder with an inlet or an exhaust port controlled by the engine valve control apparatus as claimed in any one of the preceeding claims.

13. An internal combustion engine as claimed in Claim 12 having at least one actuator connected to at least one inlet valve and having a user operable control element for controlling engine speed and a sensor for generating a signal indicative of desired

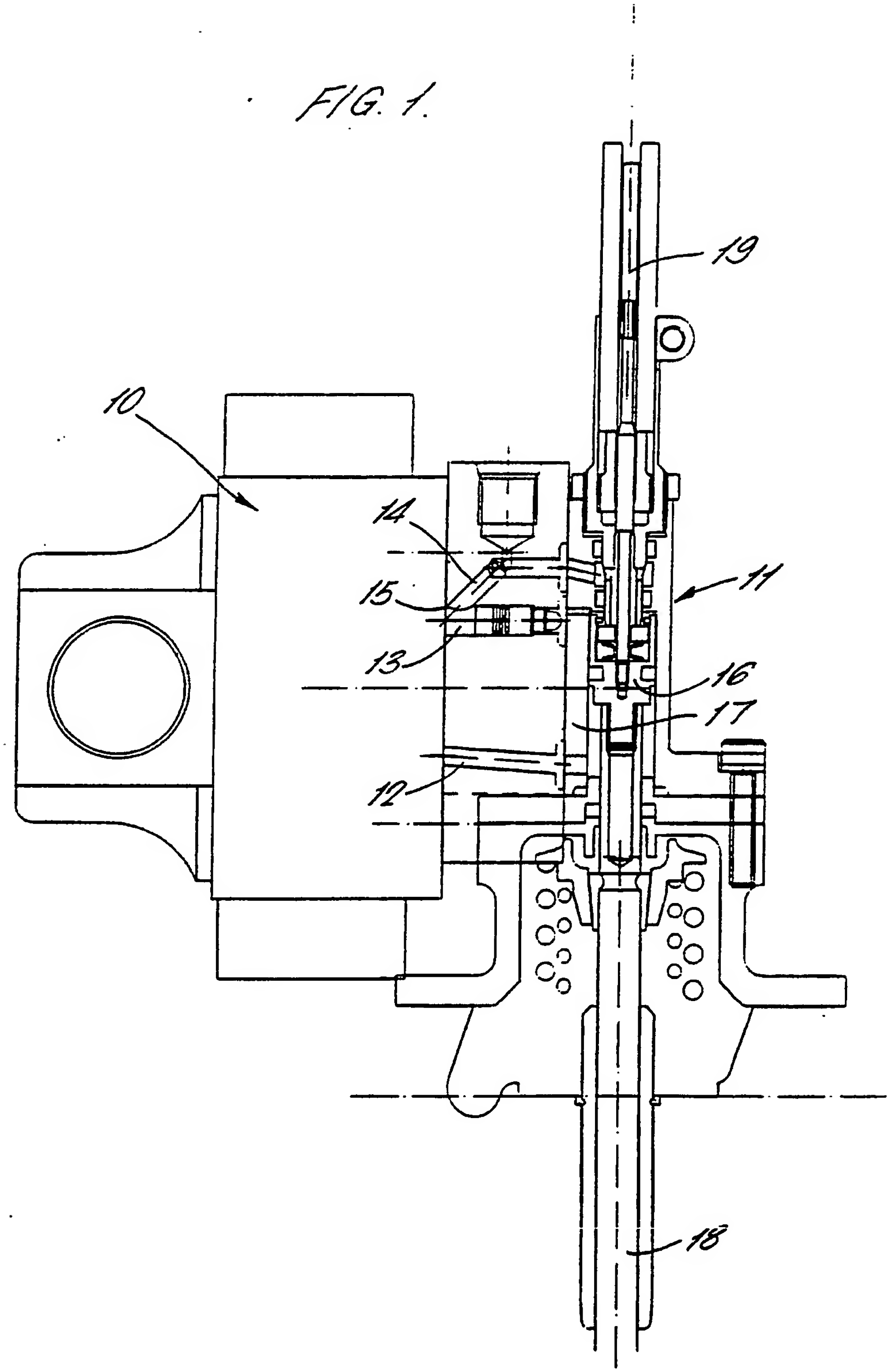
engine speed, wherein the control means receives the desired engine speed signal and controls the actuator to vary the position of the inlet valve to effect throttling of the engine.

14. A vehicle having an engine as claimed in Claim 12 or Claim 13.

15. Valve control apparatus substantially as hereinbefore described with reference to the accompanying drawings.

16. An internal combustion engine having engine valve control apparatus substantially as hereinbefore described with reference to the accompanying drawings.

FIG. 1.



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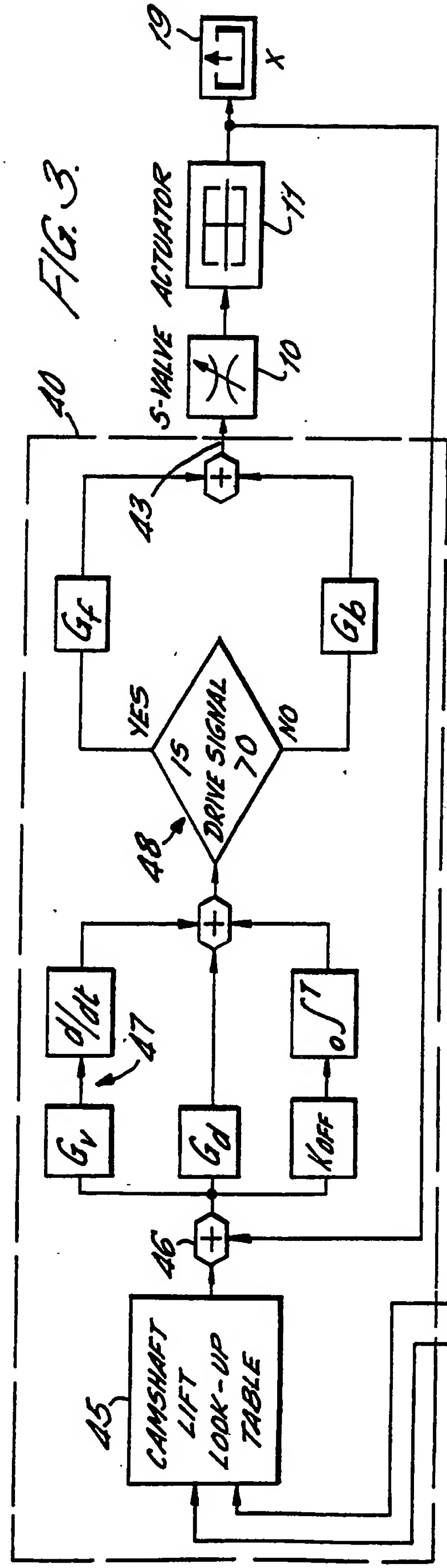

19
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ACTUATOR
11
S-VALVE
10
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43
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YES
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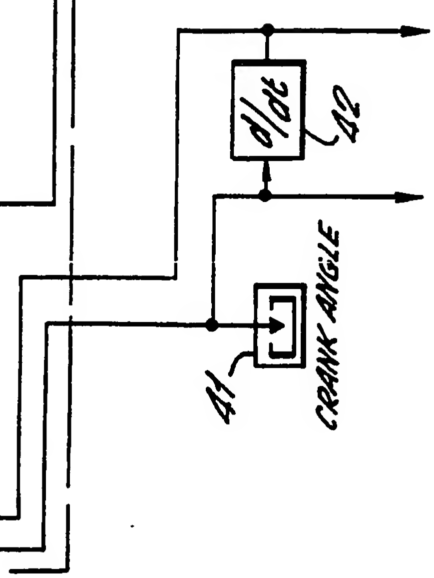

42
d/dt
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CRANK ANGLE

FIG. 3.

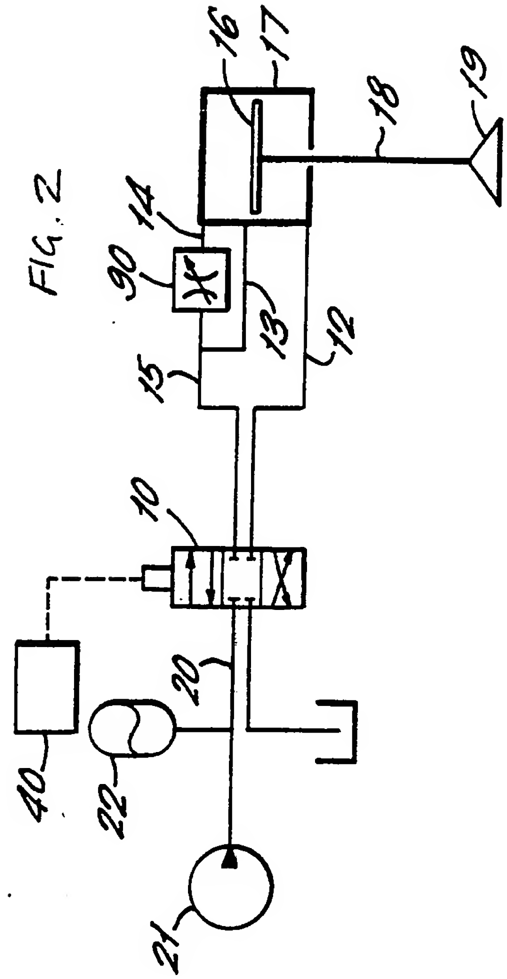

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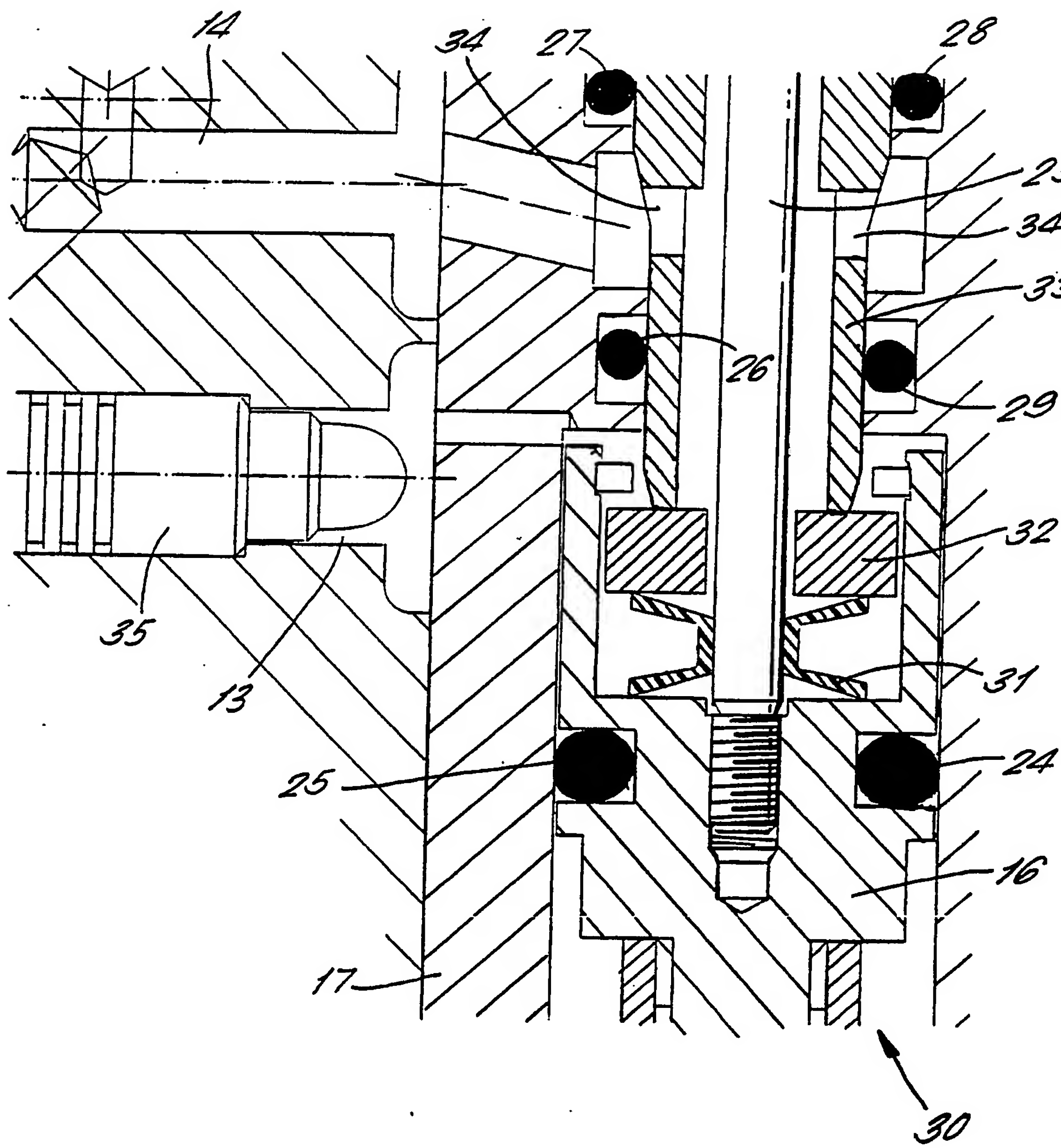
FIG. 2

FIG. 4.



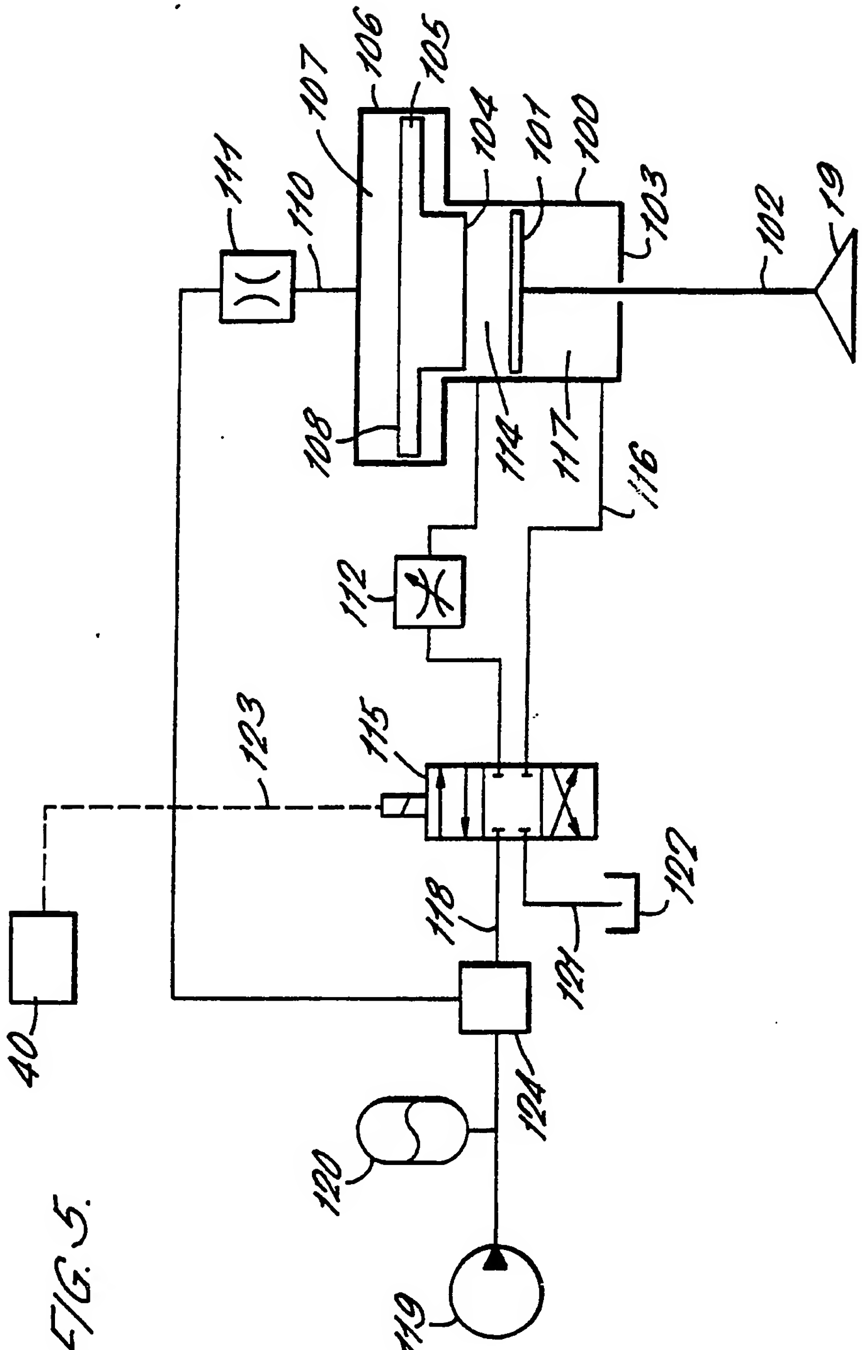

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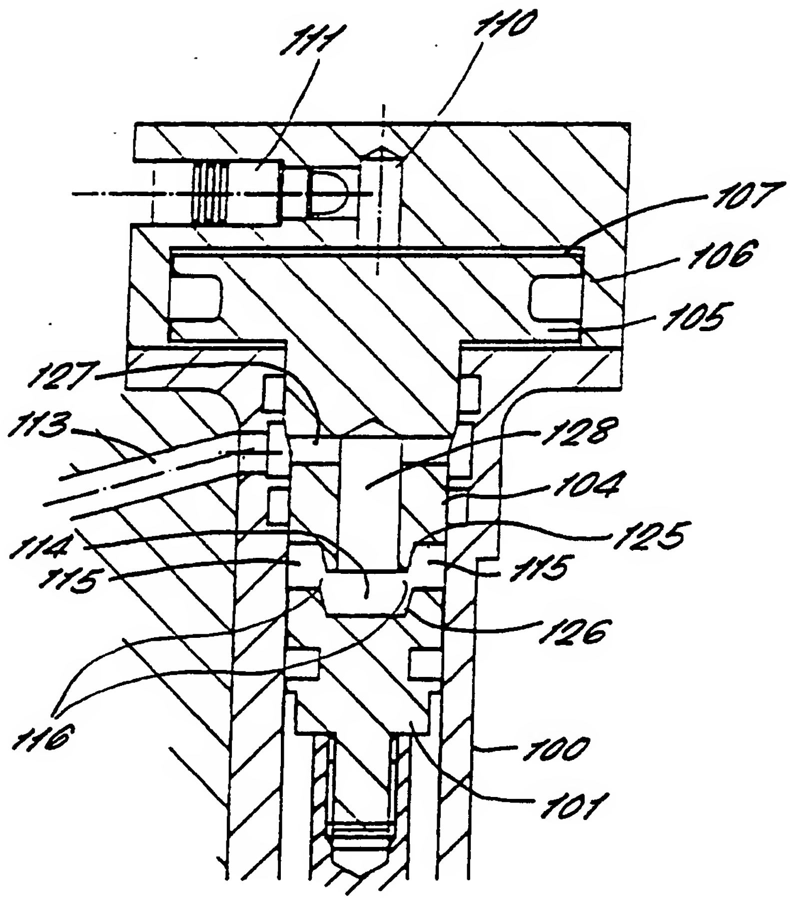
FIG. 5.

FIG. 6.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NTERNATIONAL SEARCH REPORT

International Application No PCT/GB 91/01802

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all)[6]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int.Cl. 5 F01L9/02

II. FIELDS SEARCHED

Minimum Documentation Searched[7]

Classification System	Classification Symbols
Int.Cl. 5	F01L

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched[8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT[9]

Category °	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No.[13]
X	EP,A,0 143 128 (MOLONEY) 5 June 1985 see page 5, line 7 - page 7, line 18 see page 7, line 34 - page 8, line 2 see figures 2,3,5 ---	1-16
X	US,A,3 682 152 (MÜLLER-BERNER) 8 August 1972 see column 3, line 48 - column 4, line 67 see column 5, line 45 - line 56 see figure 2 ---	1-4,6-16
P,X	US,A,4 974 495 (RICHESON) 4 December 1990 see abstract; figures 1,12 ---	1-16
X	PATENT ABSTRACTS OF JAPAN vol. 9, no. 25 (M-355)(1748) 2 February 1985 & JP,A,59 170 414 (NISSAN) 26 September 1984 see abstract ---	1,5

° Special categories of cited documents : [10]

"A" document defining the general state of the art which is not considered to be of particular relevance

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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
17 FEBRUARY 1992	24. 02. 92
International Searching Authority	**Signature of Authorized Officer**
EUROPEAN PATENT OFFICE	LEFEBVRE L.J.F.

ANNEX TO THE INTERNATIONAL SEARCH REPORT ON INTERNATIONAL PATENT APPLICATION NO. GB 9101802 SA 52288

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report.
The members are as contained in the European Patent Office EDP file on
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Patent document cited in search report	Publication date	Patent family member(s)	Publication date
EP-A-0143128	05-06-85	None	
US-A-3682152	08-08-72	DE-A- 1916167 GB-A- 1277989	15-10-70 14-06-72
US-A-4974495	04-12-90	EP-A- 0443218	28-08-91

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For more details about this annex : see Official Journal of the European Patent Office, No. 12/82